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ANGULAR BALL BEARING
[Angyura tama jikuuke]

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Claim

In an angular ball bearing provided with an inner ring that has an inner ring groove in the outer circumferential face and is attached to a rotating shaft, an outer ring that has an outer ring groove in the inner circumferential face and is disposed coaxially to house the aforementioned inner ring, multiple balls that roll between the aforementioned inner ring groove and the aforementioned outer ring groove, a cage that holds the aforementioned balls, and a groove relief part formed in the inner circumferential face of the aforementioned outer ring; an angular ball bearing characterized by the fact that the aforementioned groove relief part has a tapered curved surface shape or a tapered shape that opens out, and the shape of the outer circumferential face of the aforementioned cage at the location of the aforementioned groove relief part is formed complementarily with the shape of the inner circumferential face of the aforementioned outer ring.

Detailed explanation of the invention

[0001]

Technical field of the invention

The present invention relates to an angular ball bearing, and in particular to an angular ball bearing with which the occurrence of vibration and noise can be prevented in the cage.

[0002]

Prior art

Angular ball bearings known conventionally are bearings that support a thrust load in addition to a radial load.

[0003]

Figure 10 is a cross section of a conventional angular ball bearing in the direction of thrust, and Figure 11 is a cross section at line XI-XI in Figure 10.

[0004]

Angular ball bearing 100 has inner ring 101 attached to a rotating shaft, which is not shown, outer ring 102 placed coaxially so as to house inner ring 101, twelve balls 103 that roll between inner ring 101 and outer ring 102, and cage 104 that holds balls 103.

[0005]

An inner ring groove 105 with an arc-shaped horizontal cross section is formed in the outer circumferential face of inner ring 101, and outer ring groove 106 with a an arc-shaped horizontal cross section is formed in the inner circumferential face of outer ring 102. Multiple balls 103 are arranged between inner ring groove 105 and outer ring groove 106. Balls 103 contact inner ring groove 105 at point P and outer ring groove 106 at point Q with contact angle α (Figure 10). In the inner circumferential face of outer ring groove 106, groove relief part 107 is formed with its internal diameter increasing near the depth of outer ring groove 106 on the point P side where ball 103 contacts inner circumferential groove 105, that is, at one end of bearing 100, relative to the direction of thrust of bearing 100 (hereafter simply called "direction of thrust")

[0006]

Insertion of the twelve balls 103 held in cage 104 between inner ring groove 105 and outer ring groove 106 is accomplished by pushing them in through groove relief part 107 from one end of bearing 100 with

outer ring 102 expanded using high-frequency heating. Because of this, balls 103 will not come away from between inner ring groove 105 and outer ring groove 106 by the contraction of outer ring 102 when the temperature of outer ring 102 drops to normal temperature. In an axial [sic; angular] ball bearing 100 such as this, balls 103 revolve and rotate on inner ring groove 105 and outer ring groove 106 with relative rotation of inner ring 101 and outer ring 102.

[0007]

Figure 12 is an oblique view of cage 104.

[0008]

Cage 104 forms a tube shape that can be housed between inner ring 101 and outer ring 102, and twelve round pockets 108 are formed at equal angular intervals θ (Figure 11) in the circumferential orientation. Diameter dc of pockets 108 is set larger than diameter db of balls 103 by diametral clearance δc (= dc–db), for example, 0.6-0.7 mm (Figure 11).

[0009]

In Figure 10, a clearance for angular ball bearing 100 lubrication is furnished between the inner circumferential face of cage 104 and the outer circumferential face of inner ring 101, and the external diameter of cage 104 is set smaller than the internal diameter of outer ring 106 by diametral clearance δa, for example, 0.75-0.9 mm.

[0010]

With a construction as above, balls 103 are held at equal angular intervals θ by cage 104, inner ring groove 105 and outer ring groove 106. The position/bearing of cage 104 is controlled according to the position, in the circumferential direction, and diametral clearance θa of balls 103, as shown in Figure 13, relative to the radial direction of bearing 100 (simply called "radial direction["] hereafter), according to the position, in the direction of thrust, and diametral clearance δc of balls 103, relative to the direction of thrust.

[0011]

Problems to be solved by the invention

However, at the other end of bearing 100, with the slide bearing formed between the outer circumferential face of cage 104 and the inner circumferential face of outer ring 102, because of the narrow width, moment rigidity is small, and tilt and positioning in the direction of thrust of cage 104 cannot be controlled. The tilt angle and position in the direction of thrust of cage 104 are unstable, and in addition, the position in the radial direction is also unstable in the transitional periods when bearing 100 starts to rotate or stops.

[0012]

Also, as shown in Figure 10, the support part on the inner circumferential face of outer ring 102 that supports cage 104 is only at the other end of bearing 100, the area of this support part is small, and the center of gravity of cage 104 (center of width of cage 104) overhangs the support part. For this reason, a moment produced by centrifugal force is generated due to deflection in the position of the center of gravity of cage 104, and the moment produced by centrifugal force causes deflection of the position of the center

of gravity of cage 104. Based on the fact that centrifugal force increases at twice the speed of rotation, this is particularly disadvantageous when bearing 100 is rotating at high speed, when rotation increases, and when rotation is falling, and not only is there noise, but vibration occurs, rotational precision is degraded, all of which are undesirable.

[0013]

With angular ball bearing 100 as described above, cage 104 is supported only by the other end of bearing 100 against outer ring groove 106, that is, the center of gravity of cage 104 overhangs relative to the support part of outer ring 102 that supports cage 104. Because diametral clearance δc and diametral clearance δa are large (with an angular ball bearing with internal diameter of 100 mm, external diameter of 150 mm and width of 24 mm, δc = 0.6-0.7 mm, δa = 0.75-0.9 mm), during operation of angular ball bearing 100, the position/bearing of cage 104 will be unstable. For this reason, the position/bearing of cage 104 during operation of angular ball bearing 100 will change within the clearance range, vibration noise will be loud, and a limit on the increase in rotation speed will be produced (refer, for example, to "Dynamic Movement of Cage in Angular Ball Bearings," Koyo Engineering Journal No. 146 (1994), pp. 6-14).

[0014]

The diametral clearance δa between the internal diameter of outer ring 102 and the external diameter of cage 104 is reduced by an increase in centrifugal force along with an increase in rotation speed, and by thermal expansion of cage 104 caused by increased temperature (because it is greater than outer ring 102), so these set values can be made smaller.

[0015]

Concerning diametral clearance δc of pockets 108, because pockets 108 are simple circles, a size large enough only that flow of lubricant is not hindered from the inside of cage 104 to the outside through pockets 108 is required. The diametral clearances δa and δc cause instability in the position/bearing of the cage, and in fields where tools are frequently replaced, such as in the main spindles in the newest machining centers, and that furthermore require that rotation and stoppage be of the shortest time in order to reduce replacement time, the problems are significant. And not only instability in the position/bearing of cage 104, but in addition, lubrication of the rolling parts of balls 103, and of the sliding parts between cage 104 and balls 103, and between the outer circumferential face of cage 104 and the inner circumferential face of outer ring 102, will become basic problems in making bearing 100 faster.

[0016]

On the other hand, as the lubrication method for bearing 100, as shown in Figure 10, a lubricant such as oil mist or oil air is supplied between the inner circumferential face of cage 104 and the outer circumferential face of inner ring 101 (Q0), but because of centrifugal force acting on balls 103, a large amount [of lubricant] will flow to between the inner circumferential face of cage 104 and the outer circumferential face of inner ring 102 (Q2) at one end of bearing 100 more than between outer ring 102 and balls 103 (Q1) that are intended to be [equally] lubricated, so effective lubricating effects are hindered.

[0017]

So, the objective of the present invention is to provide an angular ball bearing with which vibration and noise in the cage can be prevented by stabilizing the position/bearing of the cage.

[0018]

Means to solve the problems

To accomplish the aforementioned objective, the angular ball bearing described in Claim 1 is characterized in that, in an angular ball bearing provided with an inner ring that has an inner ring groove in the outer circumferential face and is attached to a rotating shaft, an outer ring that has an outer ring groove in the inner circumferential face and is disposed coaxially to house the aforementioned inner ring, multiple balls that roll between the aforementioned inner ring groove and the aforementioned outer ring groove, a cage that holds the aforementioned balls, and a groove relief part formed in the inner circumferential face of the aforementioned outer ring, the aforementioned groove relief part has a tapered curved surface shape or a tapered shape that opens out, and the shape of the outer circumferential face of the aforementioned cage at the location of the aforementioned groove relief part is formed complementarily to the shape of the inner circumferential face of the aforementioned outer ring.

[0019]

With the angular ball bearing described in Claim 1, the groove relief part formed in the inner circumferential face of the outer ring is a tapered curved surface shape or a tapered shape that opens out, and the outer circumferential face of the cage at the location of the groove relief part is formed complementarily to the shape of the inner circumferential face of the outer ring. So the position of the cage can be controlled in the radial direction and the axial direction, and center of gravity position of the cage can be placed in a position where the outer circumferential face of the cage and the inner circumferential face of the outer ring are facing, and the bearing of the cage can be controlled, with the result that vibration and noise in the cage can be prevented.

[0020]

The clearance between the inner circumferential face of the cage and the outer circumferential face of the inner ring at the location of the groove relief part is also preferably larger than the clearance between the outer circumferential face of the cage and the inner circumferential face of the outer ring. The cage can be guided by the outer ring because of this.

[0021]

A V-type herringbone groove is also preferably furnished in the outer circumferential face of the cage at the location of the groove relief part. The rigidity and damping characteristics of the outer circumferential face of the cage and the inner circumferential face of the outer ring can be improved by this due to the dynamic pressure effect in the herringbone groove. The amount of lubrication between the balls and the ring grooves can also be ensured by improving the flow of lubricant, which tends to be greater on the inner ring side and less on the outer ring side.

[0022]

Even more preferably, the V-type herringbone groove should be asymmetrical. Lubricant flow from the long side of the groove to the short side is produced by this, so lubricating oil characteristics can be improved.

[0023]

Pockets to the cage are preferably furnished inclined at an angle corresponding to the contact angle of the angular ball bearing. Because of this, the contact points of the balls with the cage can be positioned on the axis of revolution of the balls, with the result that abrasion and wear in the bearing can be reduced.

[0024]

A predetermined clearance is also preferably made on the side of the bearing with the groove relief part, and a spacer is furnished on the rotating shaft facing the inner ring and the cage. By appropriately setting the clearance between the end face of the cage and the spacer, the quantity of lubricant discharged from between the inner circumferential face of the outer ring and the outer circumferential face of the cage, and the amount discharged from between the outer circumferential face of the inner ring and the inner circumferential face of the cage can be adjusted to the proper ratio, relative to the amount supplied to the bearing.

[0025]

The manufacturing method for the aforementioned cage may include an injection molding process to produce a basic ring by injection molding with high-strength plastic, a carbon fiber wrapping method to wrap carbon fibers on the outer circumferential face of said basic ring that is produced, an insert molding process to insert mold a V-type herringbone groove and stepped part with a predetermined angle of inclination using highly-lubricating plastic, and a machine finishing process to machine finish circular pockets in the ring that has been insert molded.

[0026]

With a cage that is manufactured with these processes, expansion due to centrifugal force and expansion due to increased temperature can be prevented by the carbon fiber reinforcement, and this is also beneficial against slippage produced by high rotation and stopping. Abrasion and wear produced by contact between the outer circumferential face of the cage and the inner circumferential face of the outer ring can be prevented by insert molding with highly-lubricating plastic on the outer circumference after wrapping with carbon fibers. In addition, with the combination of the aforementioned high-strength plastic, carbon fibers and highly-lubricating plastic, there is a tendency for the clearance between the outer circumferential face of the cage and the inner circumferential face of the outer ring to become smaller, since the coefficient of linear expansion is larger than the steel of the inner ring and outer ring with increased temperature, but for centrifugal force, since the density is lower than steel, there is a tendency for the clearance between the outer circumferential face of the cage and the inner circumferential face of the outer ring to become larger, and said clearance for guiding the cage can be optimized.

[0027]

Bypasses are also preferably furnished at the four corners of the circular pockets. The flow of lubricating oil can be improved through the bypasses, and the cooling and lubricating effects of the lubricating oil can be improved. In addition, the contact area between the balls and the cage can be reduced, abrasion can be reduced, and heat generation can be suppressed.

[0028]

Embodiments of the invention

An angular ball bearing pertaining to a first embodiment of the present invention is explained below while referring to figures.

[0029]

Figure 1 is a cross section, in the direction of thrust, of an angular ball bearing pertaining to a first embodiment of the present invention.

[0030]

Angular ball bearing 10 is provided with inner ring 11 that is attached to a rotating shaft, which is not shown, outer ring 12 disposed coaxially so as to house inner ring 11, multiple, for example, twelve, balls 13 that turn between inner ring 11 and outer ring 12, and cage 14 that holds balls 13. The number of balls 13 is set larger the more rigidity is required for the bearing, such as an angular ball bearing used for a high-speed spindle of a machine tool, for example.

[0031]

Inner ring groove 15 with an arc-shaped horizontal cross section is formed in the outer circumferential face of inner ring 11, and outer ring groove 16 with an arc-shaped horizontal cross section is formed in the inner circumferential face of outer ring 12. Multiple balls 13 are arranged between inner ring groove 15 and outer ring groove 16. Balls 13 contact inner ring groove 15 at point P, and outer ring groove 16 at point Q with contact angle α, for example, 15°. Groove relief part 17 is formed with its internal diameter increasing near the depth of outer ring groove 16 at point P where balls 13 touch inner circumferential

groove 15, that is, at one end of bearing 10. Groove relief part 17 is formed in a tapered shape that opens out at angle of inclination γ. Groove relief part 17 could also be formed in a tapered curved surface shape. Here, tapered curved surface refers to where a conical surface forms a concave surface such that angle of inclination γ becomes smaller in positions toward the outward opening. γ is normally selected within a range of 2-10º. When less than 2º, control in the axial orientation becomes weak, and when 10º is exceeded, the width of the bearing end becomes too high.

[0032]

Insertion of the multiple, for example, twelve, balls 13 held in cage 14 between inner ring groove 15 and outer ring groove 16 is accomplished by pushing in through groove relief part 17 from one end of bearing 10 with outer ring 12 expanded using high-frequency heating. Because of this, balls 13 will not come away from between inner ring groove 15 and outer ring groove 16 due to the contraction of outer ring 12 when the temperature of outer ring 12 drops to normal temperature. In an axial [sic] ball bearing 10 such as this, balls 13 revolve and rotate on inner ring groove 15 and outer ring groove 16 with relative rotation by inner ring 11 and outer ring 12.

[0033]

A variation of cage 14 is explained below referring to Figure 2 and Figure 3.

[0034]

Figure 2 is an oblique view of the variation of cage 104, Figure 3 (a) is a partial plan view of cage 14 in Figure 2, and Figure 3 (b) is a partial side view of cage 14 in Figure 2.

[0035]

Cage 14 forms a tubular shape that can be housed between inner ring 11 and outer ring 12, and has twelve round pockets 18 at equal angular intervals in the circumferential direction. Diameter dc of pockets 15 [sic; 18] is set larger by diametral clearance δc (= dc – db), for example. 0.6-0.7 mm, than diameter db of balls 13 (Figure 1).

[0036]

The shape of the portion of the outer circumferential face of cage 14 corresponding to groove relief part 17 is formed complementarily with the shape of the inner circumferential face of outer ring 12 inclined at angle of inclination γ, and presents a stepped shape in contrast to the other end of bearing 10 parallel with the inner circumferential face of outer ring 12. In this case, at the other end of bearing 10, the external diameter of cage 14 is set smaller than the internal diameter of outer ring 12 by diametral clearance δa, for example, 0.75-0.9 mm.

[0037]

The clearance between the inner circumferential face of cage 14 and the outer circumferential face of inner ring 11 at the location of groove relief part 17 should be larger than the clearance between the outer circumferential face of cage 14 and the inner circumferential face of outer ring 12. Because of this cage 14 can be guided by outer ring 12. Space for lubrication of angular ball bearing 10 can also be ensured in the clearance between the inner circumferential face of cage 14 and the outer circumferential face of inner ring 11, and lubricating oil, such as oil mist or oil air, can be supplied to bearing 10 through said clearance at the other end of bearing 10 (Figure 1).

[0038]

With a construction as described above, balls 13 are held at equal angular intervals by cage 14, inner ring groove 15 and outer ring groove 16, and the position/bearing of cage 14 is controlled according to the direction of thrust and diametral clearance δc of balls 13 relative to the direction of thrust of the bearing, and according to diametral clearance δa and the position, in the circumferential direction, of balls 13, relative to the radial direction of the bearing.

[0039]

With this embodiment, cage 14 has a step shaped part at one end of bearing 10, so center of gravity position G (distance L1 from the end face at one end of bearing 10) of cage 14 relative to the direction of thrust can be positioned within the range of width (L2), in the direction of thrust, of the support part of the inner circumferential face of outer ring 12 that supports the stepped part of cage 14. Because of this, the moment produced by centrifugal force due to deflection of center of gravity position G of cage 14 can be prevented.

[0040]

The outer circumferential face of the step shaped part of cage 14 is also inclined at angle of inclination γ, so as cage 14 rotates, in addition to support force in the radial direction on cage 14, support force in the direction of thrust is generated as a component force, its position in the thrust direction will be controlled in cooperation with contact with balls 13 on the opposite side, and rattling of cage 14 relative to the direction of thrust can be reduced. Tilting of cage 14 can be prevented by this.

[0041]

In this embodiment, as shown in Figure 2 and Figure 3, when V-type asymmetrical herringbone grooves 20 are furnished in the outer circumferential face of the step shaped part of cage 14, even further performance improvement is achieved by pointing the V vertices toward the opposite side of cage 14, in the direction of rotation (arrow in Figure 3). That is, a grooved bearing is constituted by clearance $\delta a/2$ between the outer circumferential face of cage 14 and the inner circumferential face of outer ring 12, and the rigidity and damping characteristics of the outer circumferential face of cage 14 and the inner circumferential face of outer ring 12 can be improved by the dynamic pressure effect of the herringbone grooves.

[0042]

Herringbone grooves 20 have width b1 and depth k1, and the vertex V is formed at a position that divides the width of the support part of cage 14 into two – width B1 and width B2 (B1 > B2). Because the V-type herringbone grooves are asymmetrical in this way (B1 > B2), a flow of lubricant from the long side to the short side of the grooves is produced, and the amount of lubricating oil for balls 13 and outer ring groove 16 can be ensured by improving the flow of lubricant, which has a tendency to be greater on the inner ring 11 side and less on the outer ring 12 side.

[0043]

Figure 4 is a cross section in the direction of thrust of an angular ball bearing pertaining to a variation of the first embodiment of the present invention. As shown in Figure 4, at one end of bearing 10, clearance δk is made with the end face of cage 14, spacer 22 is furnished on shaft 21, and clearance δk may be adjusted. Because of this, the amount of oil mist or oil air discharged Q1 between the inner circumferential face of

outer ring 12 and the outer circumferential face of cage 14, and the amount discharged Q2 between the outer circumferential face of inner ring 11 and the inner circumferential face of cage 14 can be adjusted to the proper ratio, relative to the amount supplied Q0 at the other end of bearing 10. The result is that the lubricating effect on the rolling parts (particularly contact points between balls 13 and outer ring groove 16 that are subjected to the effect of centrifugal force) can be optimized.

[0044]

As the lubricant for bearing 10, lubricating oil and cooling air (-30-0ºC) could also be supplied separately, in place of oil mist or oil air. In this case, the lubricating oil is supplied with high-pressure direct injection. The rolling parts of bearing 10 can be effectively cooled by this, with the result that seizing of bearing 10 can be prevented.

[0045]

In the constitution above, when V-type asymmetrical herringbone grooves 20 are furnished in the outer circumferential face of cage 14, diametral clearance δc between pockets 18 and balls 18 [sic; 13] and diametral clearance δa between the outer circumferential face of cage 14 and the inner circumferential face of outer ring 12 may be set to the minimum value required, and may be around 1/5-1/10 the value of the conventional diametral clearances δd and δa. These values are the proper values, considering dynamic pressure effects.

[0046]

Next, an angular ball bearing pertaining to a second embodiment of the present invention is explained using Figure 5.

[0047]

Figure 5 (a) is a cross section, in the direction of thrust, of an angular ball bearing pertaining to a second embodiment of the present invention, and Figure 5(b) is a partial plan view of cage 14 of the angular ball bearing pertaining to the second embodiment. Note that in the explanation of this embodiment, the same symbols are assigned to the same parts as the first embodiment, and redundant explanations are omitted.

[0048]

The inner circumferential face of cage 14 expands in diameter at one end of bearing 10 corresponding to the step shaped part, and at the other end of bearing 10, the clearance between the inner circumferential face of outer ring 12 and the inner circumferential face of cage 14 is larger than the clearance between the outer circumferential face of inner ring 11 and the inner circumferential face of cage 14. Lubricant [sic; lubrication] for bearing 10 is achieved in the clearance between the inner circumferential face of outer ring 12 and the inner circumferential face of cage 14.

[0049]

In addition, pockets 18 formed in cage 14 are inclined at one side of bearing 10 at the same angle as angle of inclination α, for example, 15°. Because of this, contact points R and S of balls 13 against cage 14 can be positioned on the axis of revolution of balls 13, with the result that abrasion and wear in bearing 10 can be reduced. The reason for this is because, in contrast to slippage occurring between the balls and the cage because of the cage being contacted at positions apart from points on the axes of revolution in a conventional example, with this preferred example, this slippage can be eliminated.

[0050]

Figure 6 is a view at arrow A of a pocket 18 formed in cage 14 in Figure 5(a).

[0051]

In Figure 6, the sectional shape of pockets 18 formed in cage 14 is a shape such that the vicinity of axis of revolution 23 of ball 13 and the axis ($\pm X_0$) perpendicular to it are straight lines, and in circumferential portions other than said vicinity (four corners), a bypass Δ is formed. The flow of lubricant passing through diametral clearance δc can be improved by the bypasses Δ, and the lubrication and cooling capability of bearing 10, i.e., balls 13, can be improved. The contact area between balls 13 and cage 14 can also be reduced, so abrasion between balls 13 and cage 14 can be reduced, and heat generation can be suppressed.

[0052]

For the cross section of pockets 18, a sectional shape for pockets 18 with curvature radius Rp at the four corners can be made, and bypasses Δ can be formed at said four corners, by moving a cutter with cutter diameter Dc (= 2Rp) along cutter path 25 in Figure 6.

[0053]

In this embodiment, cage 14, as shown in Figure 7, could also have a step shaped part, or it could have a circular cross section with no enlarged diameter part formed in the inner circumferential face.

[0054]

An example of a manufacturing method for cage 14 pertaining to the embodiment above is explained below using Figure 8.

[0055]

This manufacturing method comprises an injection molding process (Figure 8(a)) to produce basic ring 30 with a grooved part in the outer circumferential face using injection molding with high-strength plastic (for example, PEEK (name: polyetheretherketone), a carbon fiber wrapping process (Figure 8(b)) to wrap carbon fibers 31 that are 30 μm or less in diameter, for example, on the grooved part in the outer circumferential face of said basic ring 30 that was produced, an insert molding process (Figure 8(c)) to insert-mold the stepped part with angle of inclination γ and V-type asymmetrical herringbone grooves using high lubricating plastic (e.g., oil-containing plastic), and a machine finishing process (Figure 8(d)) to machine finish, e.g., work pockets 18 in ring 30 that has been insert-molded.

[0056]

With cage 14 that is manufactured with these processes, expansion due to centrifugal force and expansion due to increased temperature can be prevented by the carbon fiber reinforcement, and this is also useful against slippage in bearing 10 caused by high-speed rotation and stopping.

[0057]

The outer circumferential part of cage 14 is produced with highly-lubricating plastic, so abrasion and wear due to contact between the outer circumferential face of cage 14 and the inner circumferential face of outer ring 12 can be prevented. In addition, with the combination of the aforementioned high-strength

plastic, carbon fibers and highly-lubricating plastic, because the coefficient of linear expansion is larger than the steel of inner ring 11 and outer ring 12, there is a tendency for the clearance between the outer circumferential face of cage 14 and the inner circumferential face of outer ring 12 to become smaller with increased temperature. For centrifugal force, however, because the density is smaller than the steel of inner ring 11 and outer ring 12, there is a tendency for the clearance between the outer circumferential face of cage 14 and the inner circumferential face of outer ring 12 to become larger, and optimization of said clearance for guiding cage 14 can be achieved.

[0058]

Note that this manufacturing method could also be applied to a conventional type of cage 104 and not only to cage 14 in the embodiment above. For cage 104 that is manufactured, as shown in Figure 9, basic ring 40 is produced with phenol resin, carbon fibers 41 are wrapped around basic ring 40, high lubricating plastic 42 (e.g., nylon) is additionally insert-molded on its outer circumference, and pockets 108 are formed in ring 40 that has been insert-molded. The outer circumferential part of ring 40 that has been insert-molded is guided by the inner circumferential face of outer ring 102, so its properties should be such that it is constituted with a material having low abrasion or wear, and which is satisfactory for precision finishing. The number of times carbon fibers 41 are wrapped is also preferably determined by the constraint requirements between inner ring 102 [sic; 101) and outer ring 102, and should be determined so that the difference in the coefficients of linear expansion of inner ring 101 and outer ring 102 will essentially be 0.

[0059]

Effect of the invention

As explained in detail above, with the angular ball bearing described in Claim 1, the groove relief part formed in the outer circumferential face of the outer ring is a tapered, curved surface shape or a tapered shape that opens out, and the outer circumferential face of the cage at the location of the groove relief part is formed complementarily to the shape of the inner circumferential face of the outer ring. So the position of the cage can be controlled in the radial direction and the direction of thrust. The center of gravity position of the cage is also placed at a position where the outer circumferential face of the cage and the inner circumferential face of the outer ring are facing and the bearing of the cage can be controlled, with the result that vibration and noise in the cage can be prevented.

Brief description of the figures

Figure 1 is a cross section, in the direction of thrust, of an angular ball bearing pertaining to a first embodiment of the present invention.

Figure 2 is an oblique view of a variation of cage 14.

Figure 3(a) is a partial plan view of cage 14 in Figure 2, and (b) is a partial side view of cage 14 in Figure 2.

Figure 4 is a cross section, in the direction of thrust, of an angular ball bearing pertaining to a variation of a first embodiment of the present invention.

Figure 5(a) is a cross section, in the direction of thrust, of an angular ball bearing pertaining to a second embodiment of the present invention, and (b) is a partial plan view of cage 14 in an angular ball bearing pertaining to the second embodiment.

Figure 6 is a view at arrow A of a pocket 18 formed in cage 14 in Figure 5(a).

Figure 7 is a cross section, in the direction of thrust, of a variation of an angular ball bearing pertaining to a second embodiment of the present invention.

Figure 8 is an explanatory diagram explaining the manufacturing method for cage 14; (a) represents the injection molding process, (b) the carbon fiber wrapping process, (c) the insert molding process, and (d) the machine finishing process.

Figure 9 is an explanatory diagram for the manufacturing method for cage 104.

Figure 10 is a cross section, in the direction of thrust, of a conventional angular ball bearing.

Figure 11 is a cross section at line X-X in Figure 10.

Figure 12 is an oblique view of cage 104.

Figure 13 is an explanatory diagram for cage control relative to the radial direction.

Explanation of symbols

10	Angular ball bearing
11	Inner ring
12	Outer ring
13	Ball
14	Cage
15	Inner ring groove
16	Outer ring groove
17	Groove relief part
18	Pocket
20	Herringbone groove

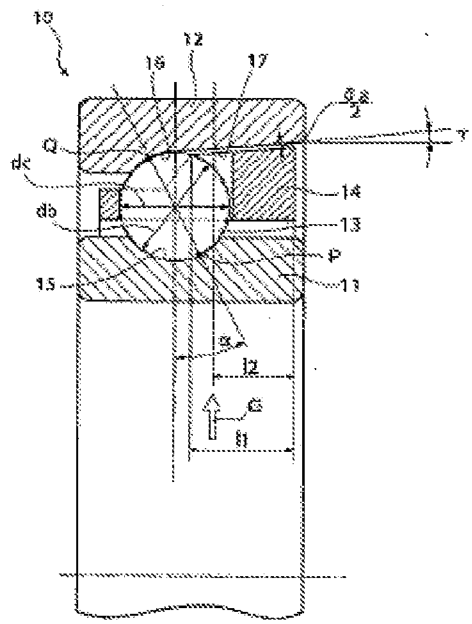

Figure 1

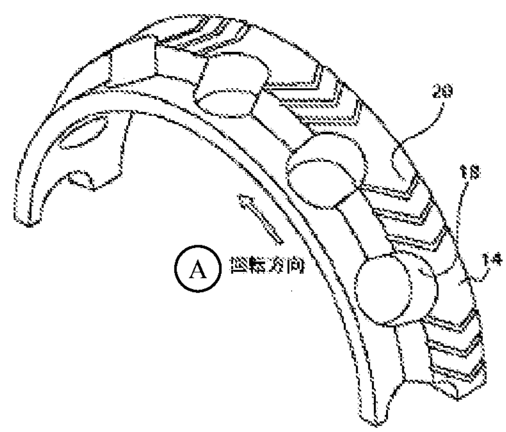

Figure 2

Key: A Direction of rotation

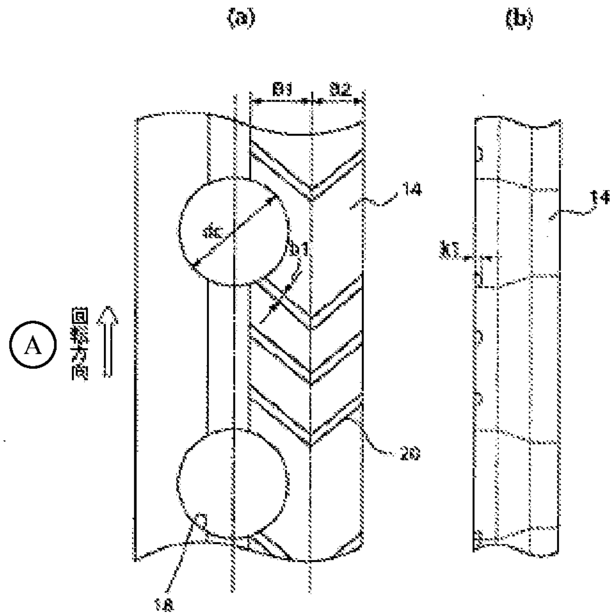

Figure 3

Key: A Direction of rotation

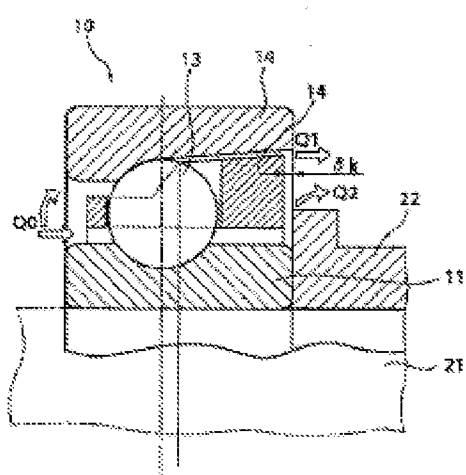

Figure 4

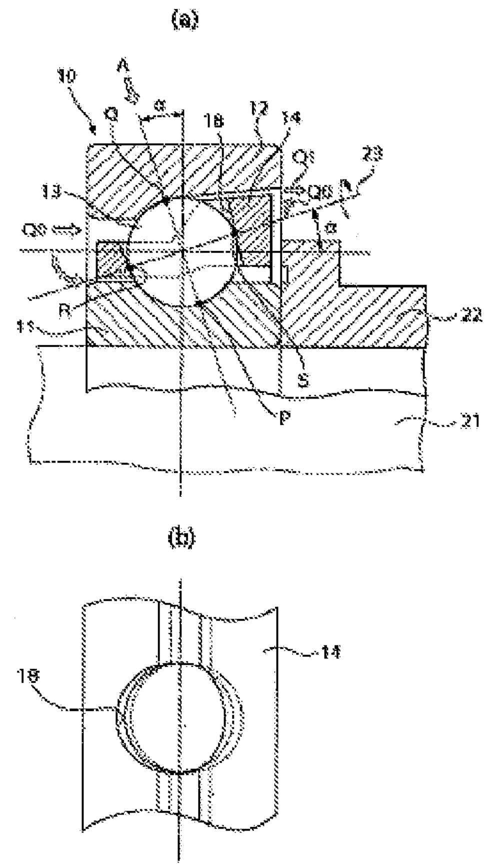

Figure 5

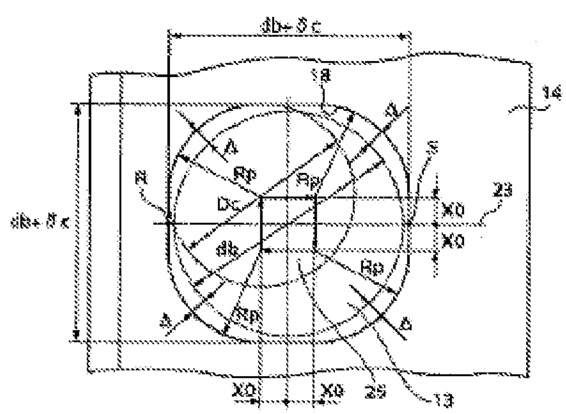

Figure 6

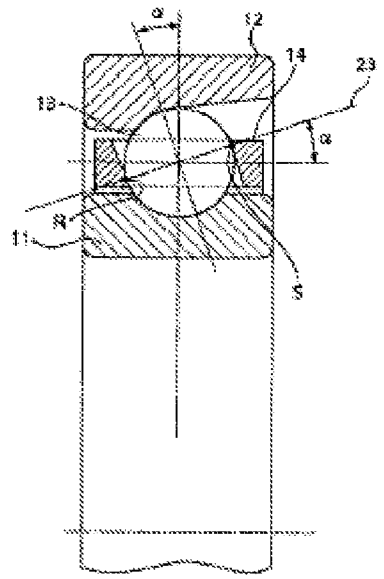

Figure 7

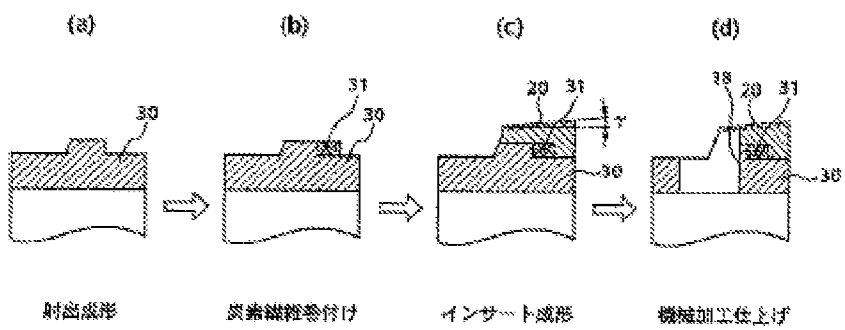

Figure 8

Key: (a) Injection molding

(b) Carbon fiber wrapping

(c) Insert molding

(d) Machine finishing

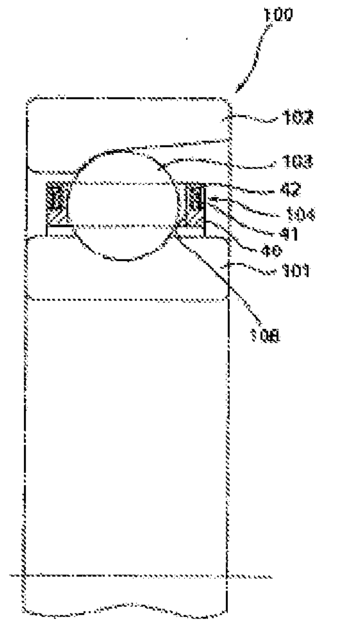

Figure 9

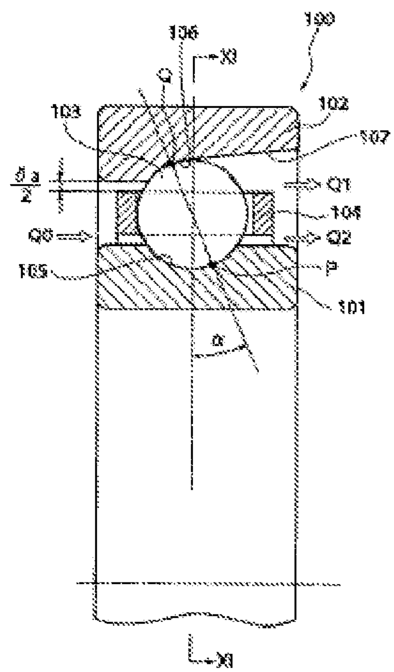

Figure 10

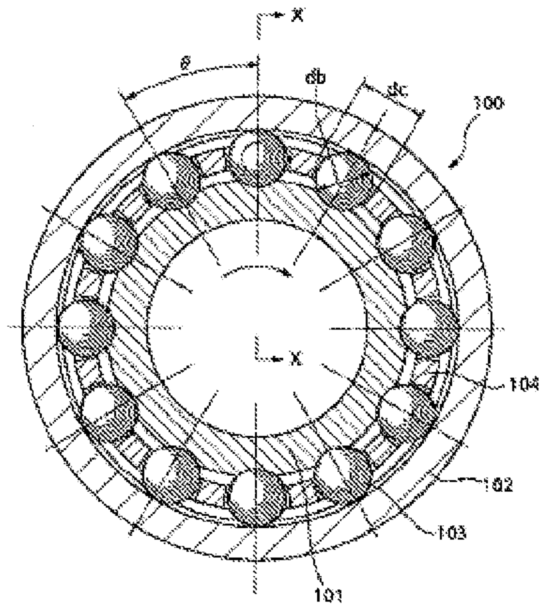

Figure 11

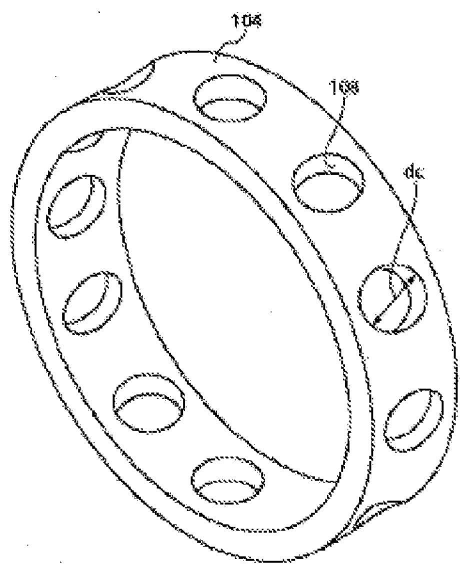

Figure 12

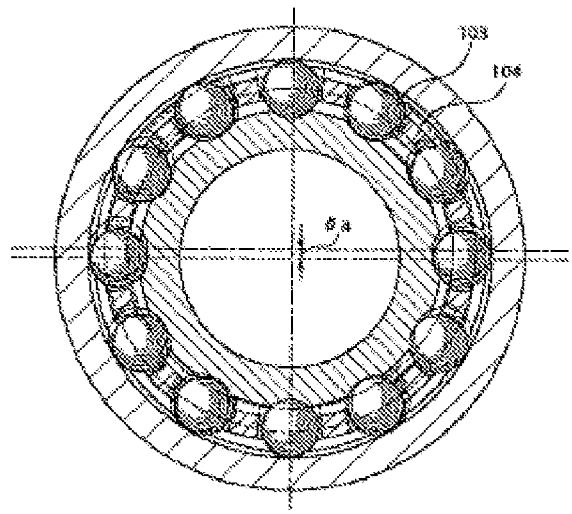

Figure 13